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Procedia Engineering 144 (2016) 1086 – 1093

Procedia Engineering

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12th International Conference on Vibration Problems, ICOVP 2015

Non-linear seventeen degrees of freedom ride dynamics model of a full tracked vehicle in simMechanics

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Abstract

Ride dynamics models may be used to study the vehicle vibration levels over various terrain conditions. In the present work, the tracked vehicle, containing 17 dof with 14 suspension stations along with an additional driver's seat bounce degree of freedom, has been developed in simMechanics. The simMechanics model incorporates the equivalent vertical nonlinear suspension characteristics of tracked vehicle suspension system, in each of the road-wheel stations. The sprung mass consists of the bounce, pitch and roll degrees of freedom and each of the unsprung masses consist of bounce degree of freedom. An additional degree of freedom is also formulated in the model, by virtue of driver seat motion. The tracked vehicle responses have been captured over defined terrains. The simMechanics model has been validated with an equivalent non-linear ride mathematical model of the full tracked vehicle. The developed simMechanics model may be used for vehicle dynamic control related studies and also may easily be co-simulated with multi-physical domains.

Peer-review under responsibility of the organizing committee of ICOVP 2015

Keywords: *Nonlinear stiffness; ride dynamics; simMechanics; governing equations.*

1. Introduction

It is essential to develop ride dynamics model of a tracked vehicle in simMechanics, in order to easily have the capability of co-simulating in multi physical domains. SimMechanics, designed as a multi body simulation tool of MATLAB software, has been used to represent dynamic systems, using predefined blocks.

SimMechanics has provisions to interface with Simscape and therefore enables direct integration with electrical, mechanical and electronic systems, thereby representing an integrated coupled multi-physical domain. A hydro- gas

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1877-7058
Peer-review under responsibility of the organizing committee of ICOVP 2015
doi:10.1016/j.proeng.2016.05.067

suspension system is modelled using polytrophic gas compression law to represent spring characteristics, while damping orifices are modelled using hydraulic conductance law by Solomon U. and Padmanabhan C. [1]. A nonlinear mathematical ride dynamic model for single station representation of tracked vehicle, has been formulated and validated with MSC. Adams by Banerjee.S, Balamurugan V. et al [2]. A methodology, optimizing the suspension kinematics of a tracked vehicle by reorienting cylinder axis of hydro gas suspension was developed by Sridhar S. and Sekar N. [3]. A ride model of a tracked vehicle in simMechanics with 17 dof and linear suspension characteristics has been modelled by Venkatasubramanian.N, Banerjee.S et.al [4]. The available literature shows extensive research undertaken in the field of tracked vehicle dynamics.

In the present work, 17 dof tracked vehicle model integrated with an additional driver's seat bounce motion, has been modelled in simMechanics and validated with Matlab. The model contains vertical nonlinear suspension stiffness characteristics of a tracked vehicle suspension system.

Nomenclature

M	sprung mass	x_{il}	bounce motion of left unsprung mass
m_{il}	left station sprung mass	x_{ir}	motion right unsprung mass
m_{ir}	right station sprung mass	K_{il}	left station suspension stiffness
m_s	driver seat mass	K_{ir}	right station suspension stiffness
I_p	moment of inertia about pitch axis	K_s	driver seat stiffness
I_r	moment of inertia about roll axis	C_{il}	left station suspension damping
Z	bounce motion of sprung mass	C_{ir}	right station suspension damping
x_s	bounce motion of driver seat	K_{tli}	left station tyre stiffness
θ	pitch motion of sprung mass	K_{tri}	right station tyre stiffness
ϕ	roll motion of sprung mass		
l_{il}	distance between vehicle CG and left wheel station along pitch axis		
l_{ir}	distance between vehicle CG and right wheel station along pitch axis		
a	distance between vehicle CG and left wheel station along roll axis		
b	distance between vehicle CG and right wheel station along		
e_p	distance between driver seat & vehicle CG along pitch axis		
e_r	distance between driver seat & vehicle CG along roll axis		

2. Ride vibration model development in simMechanics

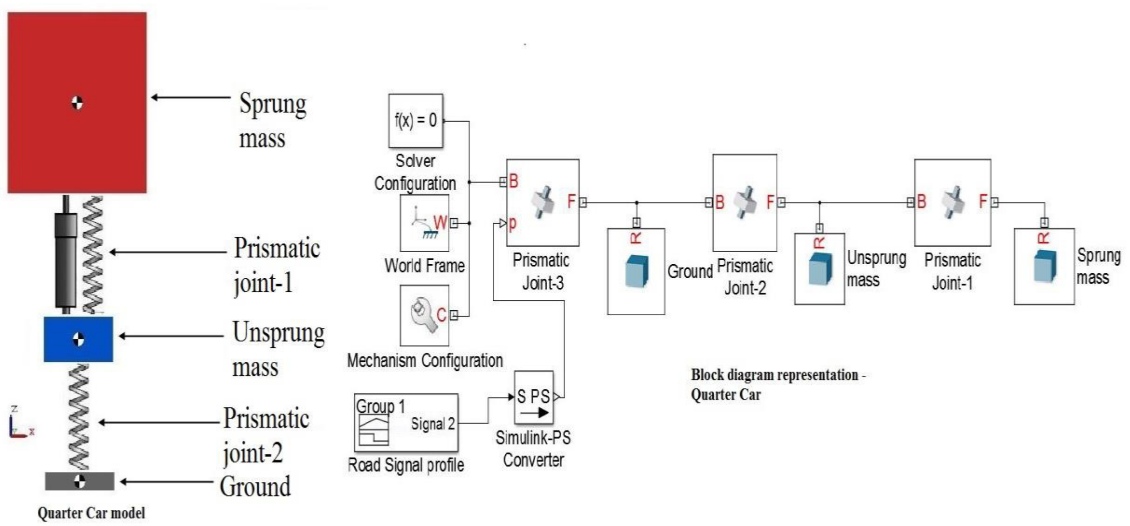


Fig -1 **(a)** Multi body representation in simMechanics - Quarter car (b) block diagram representation

2.1 Quarter Car representation of full tracked vehicle

A quarter car, modelled in simMechanics and its simMechanics block representation, has been shown in Figure -1. Sprung and unsprung masses have been created by using a solid block. The inertial properties of the bodies may be specified in the solid block. Relative motion between the bodies may be specified by using joints block. In the quarter car model, sprung mass and unsprung mass have relative translational motion between each other, which may be provided by the prismatic joint (shown in Figure-1).

Suspension stiffness and damping properties have been specified in prismatic joint-1. In order to provide road profile as base excitation, a point mass (ground) has been created (as shown in Figure-1). Relative vertical translational motion between unsprung mass and ground is provided by prismatic joint-2. Tyre stiffness has been specified in prismatic joint-2. In the block representation, prismatic joint-3 is used to achieve vertical translational motion, as the vehicle negotiates various road profiles. The other auxiliary blocks (as shown in Figure-1), such as solver configuration block, may be used to select suitable solver for the model. Explicit solver ode 45, has been used to solve the equations of motion. The mechanism configuration block (as shown in Figure-1), is used to incorporate gravitational effects in the vibration model. Road profiles are provided as base excitation to the model using the signal builder block through Simulink to physical signal converter. The world frame block (as shown in Figure-1), is the fixed frame of reference for the model in global co-ordinates. Transform sensors may be used to measure the sprung mass and unsprung mass motions.

2.2 Development of the 17 dof tracked vehicle model with additional driver's seat bounce motion

The 17 dof tracked vehicle model with additional driver's seat bounce, as well as its equivalent block representation, have been shown in Figure-2 and Figure-3, respectively. Therefore, the dynamic model totally consists of 18 dof. The modelling approach for quarter car has been extended to develop the above model in simMechanics. Pin slot joint has been provided to achieve pitch and roll motion of sprung mass.

As shown in Figure 3, predefined blocks used for left wheel station modelling, have been masked under the block, named as left wheel track. Similarly, right wheel station and sprung mass modelling blocks have been masked under block, named as right wheel track and sprung mass respectively. The predefined block, masked under rear side connection block, are used to connect each wheel station with sprung mass.

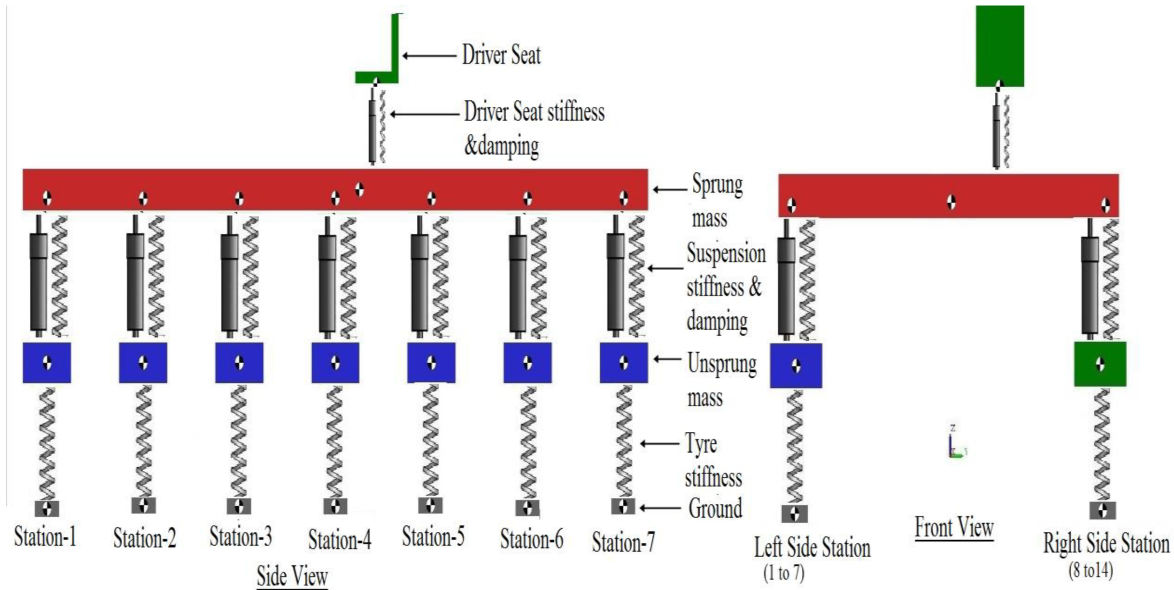


Fig-2 Multi body representation of a tracked vehicle with18 dof in SimMechanics

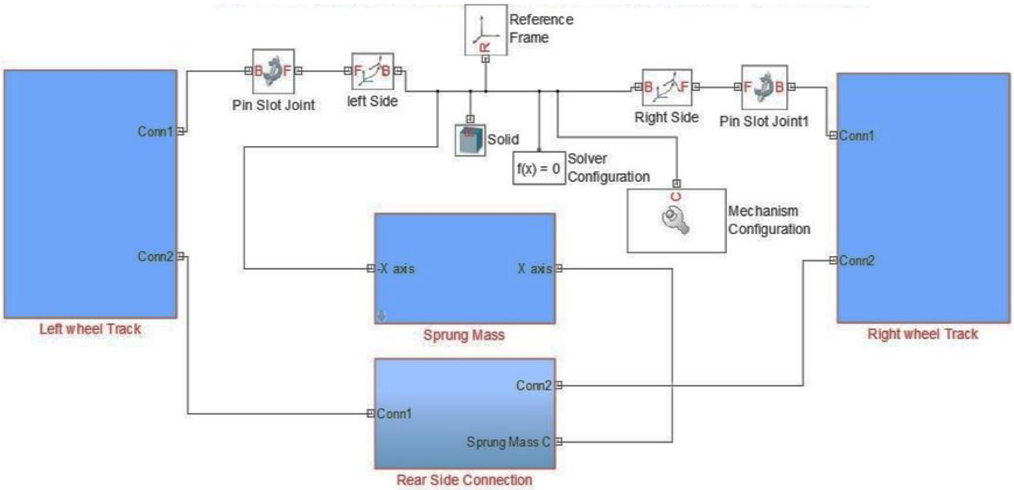


Fig-3 Block representation of 18 dof multibody model of tracked vehicle

3. Mathematical model formulation

The sprung mass responses, obtained from the simMechanics model, have been validated with an equivalent mathematical model, shown in Figure-4. The differential equations have been formulated, coded and solved in Matlab. The sprung mass contains 3 degrees of freedom, namely bounce, pitch and roll motions, whereas each of the 14 unsprung masses has bounce motion, which along driver seat's bounce motion, constitutes 18 degrees of freedom. The nomenclatures used for the model, are given in Table-1

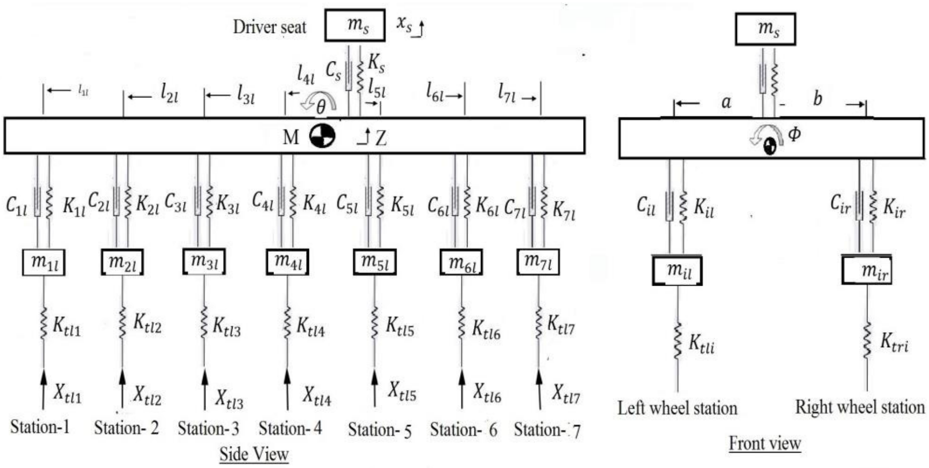


Fig 4 Mathematical model representation in global coordinates

The bounce motion of sprung mass at its CG is given by

$$M.\ddot{Z} + \sum_{i=1}^{4}[K_{il}.(Z - x_{il} - (\theta.l_{il}) - (\phi.a))] + \sum_{i=5}^{7}\left[K_{il}.\left(Z - x_{il} + (\theta.l_{il}) - (\phi.a)\right)\right] + \sum_{i=1}^{4}[K_{ir}.(Z - x_{ir} - (\theta.l_{ir}) + (\phi.b))] + \sum_{i=5}^{7}\left[K_{ir}.\left(Z - x_{ir} + (\theta.l_{ir}) + (\phi.b)\right)\right] + \sum_{i=1}^{4}[C_{il}.(\dot{Z} - \dot{x}_{il} - (\dot{\theta}.l_{il}) - (\dot{\phi}.a))] + \sum_{i=5}^{7}\left[C_{il}.\left(\dot{Z} - \dot{x}_{il} + (\dot{\theta}.l_{il}) - (\dot{\phi}.a)\right)\right] + \sum_{i=1}^{4}[C_{ir}.(\dot{Z} - \dot{x}_{ir} - (\dot{\theta}.l_{ir}) + (\dot{\phi}.b))] + \sum_{i=5}^{7}\left[C_{ir}.\left(\dot{Z} - \dot{x}_{ir} + (\dot{\theta}.l_{ir}) + (\dot{\phi}.b)\right)\right] + K_s.\left(Z - x_s + (\theta.e_p) + (\phi.e_r)\right) + C_s.\left(\dot{Z} - \dot{x}_s + (\dot{\theta}.e_p) + (\dot{\phi}.e_r)\right) = 0 \quad (1)$$

The pitch motion of sprung mass at its CG is given by

$$I_p.\ddot{\theta} + \sum_{i=1}^{4}[-K_{il}.l_{il}.(Z - x_{il} - (\theta.l_{il}) - (\phi.a))] + \sum_{i=5}^{7}\left[K_{il}.l_{il}.\left(Z - x_{il} + (\theta.l_{il}) - (\phi.a)\right)\right] + \sum_{i=1}^{4}\left[-K_{ir}.l_{ir}.(Z - x_{ir} - (\theta.l_{ir}) + (\phi.b))\right] + \sum_{i=5}^{7}\left[K_{ir}.l_{ir}.\left(Z - x_{ir} + (\theta.l_{ir}) + (\phi.b)\right)\right] + \sum_{i=1}^{4}[-C_{il}.l_{il}.(\dot{Z} - \dot{x}_{il} - (\dot{\theta}.l_{il}) - (\dot{\phi}.a))] + \sum_{i=5}^{7}\left[C_{il}.l_{il}.\left(\dot{Z} - \dot{x}_{il} + \left(\dot{\theta}.l_{il}\right) - \left(\dot{\phi}.a\right)\right)\right] + \sum_{i=1}^{4}\left[-C_{ir}.l_{ir}.\left(\dot{Z} - \dot{x}_{ir} - \left(\dot{\theta}.l_{ir}\right) + (\dot{\phi}.b)\right)\right] + \sum_{i=5}^{7}\left[C_{ir}.l_{ir}.\left(\dot{Z} - \dot{x}_{ir} + \left(\dot{\theta}.l_{ir}\right) + \left(\dot{\phi}.b\right)\right)\right] + K_s.e_p.\left(Z - x_s + \left(\theta.e_p\right) + (\phi.e_r)\right) + C_s.e_p.\left(\dot{Z} - \dot{x}_s + \left(\dot{\theta}.e_p\right) + \left(\dot{\phi}.e_r\right)\right) = 0 \tag{2}$$

The roll motion of sprung mass at its CG is given by

$$I_r.\ddot{\phi} + \sum_{i=1}^{4}[-K_{il}.a.(Z - x_{il} - (\theta.l_{il}) - (\phi.a))] + \sum_{i=5}^{7}\left[-K_{il}.a.\left(Z - x_{il} + (\theta.l_{il}) - (\phi.a)\right)\right] + \sum_{i=1}^{4}\left[K_{ir}.b.(Z - x_{ir} - (\theta.l_{ir}) + (\phi.b))\right] + \sum_{i=5}^{7}\left[K_{ir}.b.\left(Z - x_{ir} + (\theta.l_{ir}) + (\phi.b)\right)\right] + \sum_{i=1}^{4}[-C_{il}.b.(\dot{Z} - \dot{x}_{il} - (\dot{\theta}.l_{il}) - (\dot{\phi}.a))] + \sum_{i=5}^{7}\left[-C_{il}.b.\left(\dot{Z} - \dot{x}_{il} + \left(\dot{\theta}.l_{il}\right) - \left(\dot{\phi}.a\right)\right)\right] + \sum_{i=1}^{4}[C_{ir}.b.\left(\dot{Z} - \dot{x}_{ir} - \left(\dot{\theta}.l_{ir}\right) + (\dot{\phi}.b))\right] + \sum_{i=5}^{7}\left[C_{ir}.b.\left(\dot{Z} - \dot{x}_{ir} + \left(\dot{\theta}.l_{ir}\right) + \left(\dot{\phi}.b\right)\right)\right] + K_s.e_r.\left(Z - x_s + \left(\theta.e_p\right) + (\phi.e_r)\right) + C_s.e_r.\left(\dot{Z} - \dot{x}_s + \left(\dot{\theta}.e_p\right) + \left(\dot{\phi}.e_r\right)\right) = 0 \tag{3}$$

Bounce motion of left unsprung mass located in front of vehicle CG is given by

$$m_{il}.\ddot{x}_{il} + K_{il}.(x_{il} + (\theta.l_{il}) + (\phi.a) - Z) + C_{il}.\left(\dot{x}_{il} + \left(\dot{\theta}.l_{il}\right) + (\dot{\phi}.a) - \dot{Z}\right) + K_{tli}.(x_{il} - X_{tli}) = 0$$
where $(i = 1\ to\ 4)$ (4)

Bounce motion of rear left unsprung mass located in rear of vehicle CG is given by

$$m_{il}.\ddot{x}_{il} + K_{il}.(x_{il} - (\theta.l_{il}) + (\phi.a) - Z) + C_{il}.\left(\dot{x}_{il} - \left(\dot{\theta}.l_{il}\right) + (\dot{\phi}.a) - \dot{Z}\right) + K_{tli}.(x_{il} - X_{tli}) = 0$$
where $(i = 5\ to\ 7)$ (5)

Bounce motion of right unsprung mass located in front of vehicle CG is given by

$$m_{ir}.\ddot{x}_{ir} + K_{ir}.(x_{ir} + (\theta.l_{ir}) - (\phi.b) - Z) + C_{ir}.\left(\dot{x}_{ir} + \left(\dot{\theta}.l_{ir}\right) - (\dot{\phi}.b) - \dot{X}\right) + K_{tri}.(x_{ir} - X_{tri}) = 0$$
where $(i = 1\ to\ 4)$ (6)

Bounce motion of right unsprung mass located in rear of vehicle CG is given by

$$m_{ir}.\ddot{x}_{ir} + K_{ir}.(x_{ir} - (\theta.l_{ir}) - (\phi.b) - Z) + C_{ir}.\left(\dot{x}_{ir} - \left(\dot{\theta}.l_{ir}\right) - \left(\dot{\phi}.b\right) - \dot{Z}\right) + K_{tri}.(x_{ir} - X_{tri}) = 0$$
where $(i = 5\ to\ 7)$ (7)

Bounce motion of driver seat located behind the CG is given by

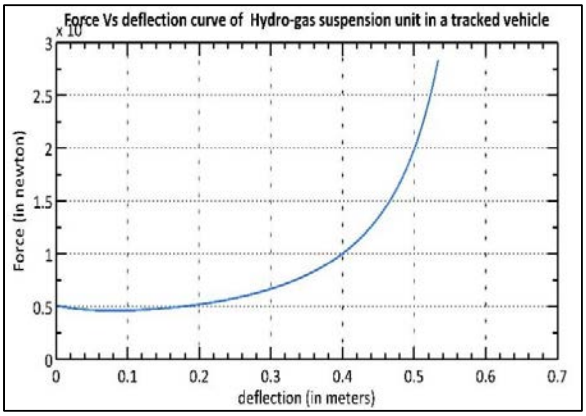
$$m_s.\ddot{x}_s + k_s.\left(x_s - \left(\theta.e_p\right) - (\phi.e_r) - Z\right) + C_s.\left(\dot{x}_s - \left(\dot{\theta}.e_p\right) - \left(\dot{\phi}.e_r\right) - \dot{Z}\right) = 0 \tag{8}$$

As shown in Figure 5 (a), force-deflection curve of a nonlinear suspension, implemented in all 14 wheel stations of the 18 dof tracked vehicle ride model, has been used for all simulation cases.
As shown in Figure 5 (b), Axle Proving Gauge (APG) terrain profile at 20 kmph is provided as base excitation at 2nd second of the simulation to the 1st wheel station. Excitation to the other wheels stations have been applied with suitable time lag, resembling a vehicle encountering APG terrain, and simulated up to 25 seconds.

Table-1 Parameters used for the simulation of 18 dof Multi body model of a tracked vehicle

Parameters	**Value**
Sprung mass	30000 kg
Unsprung mass (1 to 14)	216 kg
Pitch moment of inertia	85000 kg.m^2
Roll moment of inertia	95000 kg.m^2

Driver seat mass	100 kg
Longitudinal distance from vehicle CG to wheel station-1 & 8.	3.25 m
Longitudinal distance from vehicle CG to wheel station-2 & 9.	2.25 m
Longitudinal distance from vehicle CG to wheel station-3 & 10.	1.25 m
Longitudinal distance from vehicle CG to wheel station-4 & 11.	0.25 m
Longitudinal distance from vehicle CG to wheel station-5 & 12.	0.75 m
Longitudinal distance from vehicle CG to wheel station-6 & 13.	1.75 m
Longitudinal distance from vehicle CG to wheel station-7 & 14.	2.75 m
Lateral distance from vehicle CG to left wheel stations (1 to 7)	1.5 m
Lateral distance from vehicle CG to right wheel stations (8 to 14)	1.5 m
Suspension damping (1 to 14)	2167 Ns/m
Driver seat damping	100 Ns/m
Driver seat stiffness	25000 N/m
Tyre Stiffness	8000000 N/m



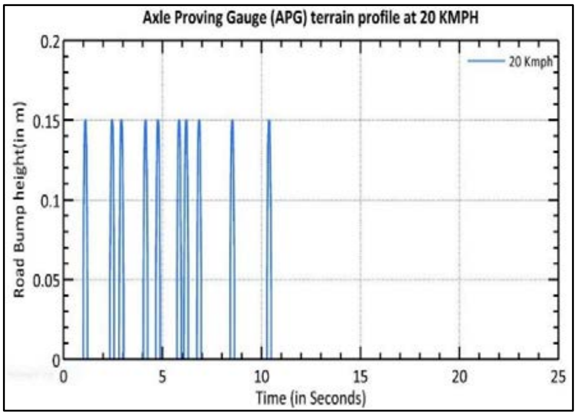
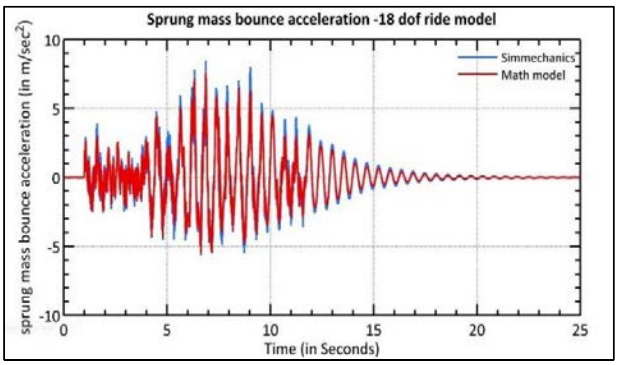


Fig – 5 (a) Force vs. deflection curve of a tracked vehicle (b) Base excitation due to APG terrain profile

4. Results and discussion

Ride model's sprung mass response, while crossing the APG terrain (as shown in Figure -5(b)) is discussed below. In Figures 6(a) and 6(b), sprung mass bounce acceleration in both time and frequency domains, obtained from simMechanics and mathematical model, are compared. It is observed that the maximum bounce acceleration magnitudes in simMechanics and mathematical model, are 8.46 m/sec^2 and 7.62 m/sec^2, respectively. The sprung mass bounce acceleration RMS values, attained from the simMechanics and mathematical models, are about 1.92 m/sec^2 and 1.74 m/sec^2, respectively. From the frequency spectrum, it may be noted that the RMS values of bounce acceleration peaks, have a close match at 2.64 Hz, from both models.



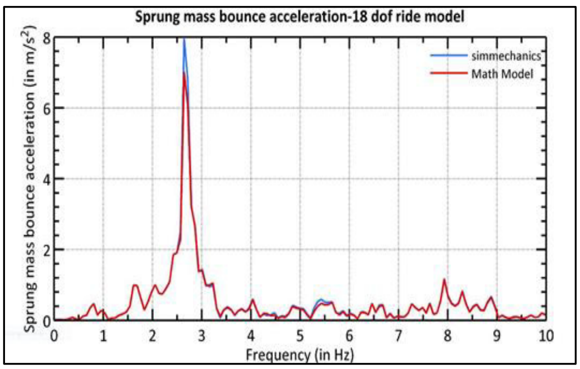
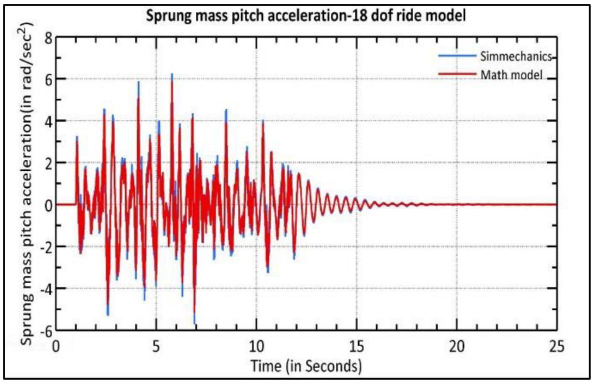


Fig-6 (a) Sprung mass bounce acceleration- time domain (b) Sprung mass bounce acceleration- frequency domain



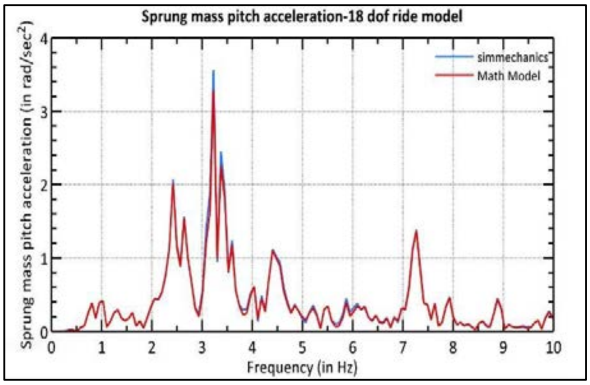
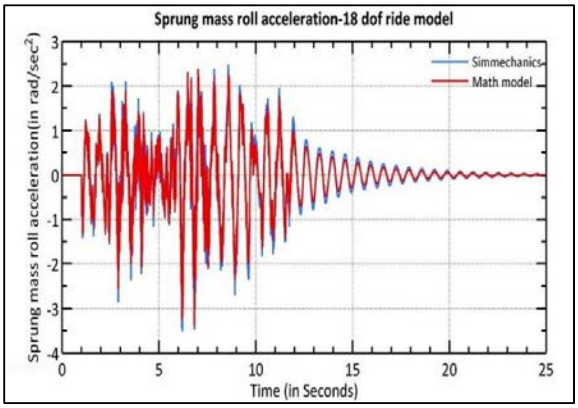


Fig-7 (a) Sprung mass pitch acceleration- time domain (b) Sprung mass pitch acceleration- frequency domain

It may be observed from Figure 7(a), that the sprung mass pitch acceleration magnitudes from simMechanics and mathematical models, are of the order of 6.26 rad/sec^2 and 5.86 rad/sec^2 respectively. Moreover, the RMS values of sprung mass pitch accelerations, observed from both models, are about 1.20 rad /sec^2 and 1.11 rad /sec^2 respectively. The frequency spectrum for sprung mass pitch acceleration (as shown in Figure 7(b)), highlights closeness of peaks at about 3.23 Hz, from both models. Sprung mass roll acceleration responses from simMechanics and mathematical models, have been compared in time domain and also in frequency domain, as shown in Figures 8(a) and 8(b), respectively. It is observed that the maximum roll acceleration magnitudes in simMechanics and mathematical model, are about 3.51 rad/sec^2 and 3.17 rad/sec^2, respectively.

It is observed from the frequency spectrum, that the roll acceleration peaks, are at 2.27 Hz in both models. The RMS value of sprung mass roll acceleration response from simMechanics and mathematical model, are 0.79 rad/sec^2 and 0.71 rad/sec^2, respectively. In Figures 9(a) and 9(b), driver's seat bounce accelerations from simMechanics and mathematical model, have been compared in both time and frequency domains, respectively. The maximum driver's seat acceleration magnitudes from simMechanics and mathematical models, are 9.66 m/sec^2 and 8.97 m/sec^2, respectively. From the frequency spectrum, the maximum driver's seat bounce acceleration values are at 2.64 Hz in both models. In Figure 10, driver's seat bounce acceleration responses from simMechanics model, at 20 kmph and 40 kmph, passing over an APG terrain, have been compared. As the vehicle approach velocity over an undulated terrain increases, the acceleration levels of driver seat will also increase. This occurrence may be observed from Figure 10, where the maximum driver's seat acceleration values at 20 kmph and 40 kmph, are 9.66 m/sec^2 and 14.34 m/sec^2, and corresponding RMS values are 2.07 m/sec^2 and 2.37 m/sec^2, respectively. The responses, obtained from the simMechanics model and math model, have quite a close match. However, little difference in response amplitudes are observed, which may be due to the solver interpolation techniques with various multi-body constraint.



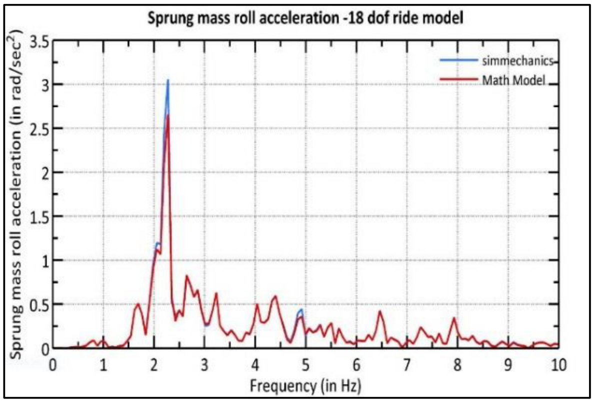


Fig-8 (a) Sprung mass roll acceleration-time domain (b) Sprung mass roll acceleration - frequency domain

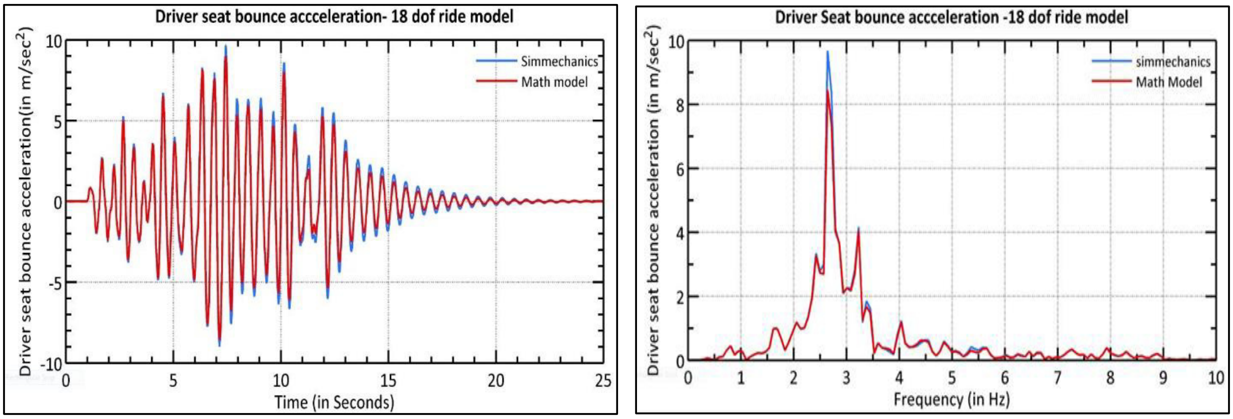


Fig 9 (a) Driver seat bounce acceleration- time domain (b) Driver seat bounce acceleration- frequency domain

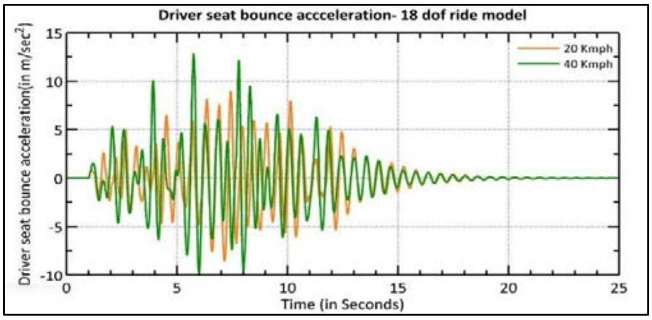


Fig-10 Driver seat bounce acceleration comparison at 20 kmph and 40 kmph

5. Conclusion

An 18 dof ride dynamics model of a tracked vehicle has been developed in simMechanics with nonlinear tracked vehicle suspension characteristics, and validated successfully with an equivalent math model in Matlab. The developed simMechanics model may be used for further vehicle dynamics control related studies. The present model has very good prospects of including the semi-active and active suspension system logic. There is also a good scope of interfacing the model with multi-physical domains, relevant to tracked vehicle suspension functionality.

Acknowledgement

The authors are grateful to Shri A. Rajadurai, Dean, Madras Institute of Technology and Shri.S.Jayaraj, Head in charge, Automobile Engg., Madras Institute of Technology, Dr. P. Siva Kumar, Director (CVRDE) and Shri. S. Ramesh, Addl. Director (CEAD), for extending all the facilities to carry out the above work. The authors also thank the CEAD team and other system groups for all their support.

References

[1] U.Solomon, Chandramouli Padmanabhan, Hydro gas suspension system of a tracked vehicle- Modeling and analysis. Journal of Terra mechanics, **48**, 125–137. (2011)

[2] Saayan Banerjee, Balamurugan.V, Ride dynamics mathematical model of a single station representation of a tracked vehicle Journal of Terra mechanics, **53**, 458. (2014)

[3] Sridhar and N.S.Sekar, Optimization of kinematics of tracked vehicle hydro gas suspension system, Defence science journal, No **5**, Vol 56. (2006).

[4] Venkatasubramanian.N, Saayan Banerjee, Balamurugan V., Seventeen degrees of freedom ride dynamics model of a tracked vehicle in simmechanics. International journal of applied engineering and research No 19, Volume 10. (2015).